Screw compressor

The present invention relates to a speed-regulated helical screw rotor compressor that is adapted to work against a pressure container whose pressure P lies within the working range of the compressor and which is allowed to vary between a lowest pressure and a highest pressure. The compressor is driven by an electric motor.

Small pressure variations are desirable in such a pressure container or accumulator tank. In the case of a large accumulator tank this can be achieved with a highly frequent start-stop control facility or by regulating the speed (r.p.m.) of the motor.

Compressor speed control is generally used in respect of air compressors that are driven by a high power motor down to a power of 10-30 kW. The compressor speed is controlled with the aid of electronic control means. With regard to compressors that are driven by weaker motors, such as motors of lower power, or much lower power, than 10-30 kW the use of compressor speed control based on electronic circuits cannot be defended economically. This is because the control electronics are extremely expensive in relation to the energy savings resulting from compressor speed control. The aforesaid lower limit of about 10-30 kW in respect of the power of a speed control compressor can, however, be lower with increased energy costs.

A smart way of controlling the pressure in the pressure container is to use in the container a pressure sensor which, via appropriate control means, functions to switch-off the compressor motor when the pressure in the container has reached its maximum value and to switch-on the motor when the container pressure has reached a pre-determined lowest value.

When using typical asynchronous motors, such control will result in rapid filling of the pressure container to a maximum pressure. When consumption is high or when the pressure container is relatively small, the motor will be frequently switched on and off. These frequent motor-starts will shorten the useful life of the motor quite considerably.

In addition to saving energy, the aim of speed control is to enable the buffer tank against which the compressor works to be made much smaller than would otherwise be the case. A compressor whose speed is not controlled will thus require a larger buffer tank and larger tank accommodating space, therewith incurring higher investment costs.

There is a need for small, speed-regulated compressors that can be used in simpler applications, such as with screwdrivers, paint sprays and various other hand tools.

Accordingly, the aim of the present invention is to provide a motor-driven compressor whose motor has a much smaller power than the aforesaid lowest power and the speed of which can be controlled at least within one working range in the absence of expensive control equipment.

This aim is achieved in accordance with the invention with a compressor that is driven by a motor whose speed is significantly dependant on the torque or moment index within a given working range. Preferred embodiments will be apparent from the dependent claims.

The present invention will now be described in more detail with reference to the accompanying drawing, in which

Figure 1 is a longitudinally sectional view of a known helical screw compressor,

Figure 2 is a sectional view taken on the line II-II in fig. 1;

Figure 3 is a diagrammatic illustration of a system which includes the compressor;

Figure 4 illustrates diagrammatically the torque of a typical compressor motor as a function of its speed (r.p.m.); and

Figure 5 is a corresponding diagrammatic illustration of a compressor motor according to the present invention.

A brief description of the construction and working principle of a helical screw compressor will now be given with reference to figures 1 and 2.

A pair of mutually engaging screw rotors 101, 102 are mounted for rotation in a working space delimited by two end walls 103, 104 and including a barrel wall 105 that extends between said end walls. The barrel wall 105 has a form which corresponds generally to that of two mutually intersecting cylinders, as evident from figure 2. Each rotor 101, 102 includes a plurality of lobes 106 and 107 respectively, and respective intermediate grooves 111 and 112 that extend in a helical line along the rotor. One rotor 101 is a male type of rotor with the major part of each lobe 106 located outside the pitch circle and the other rotor 102 is a female type rotor with the major part of each lobe 107 located inwardly of the pitch circle. The female rotor 102 will usually have more lobes than the male rotor 101. A typical combination is one in which the male rotor 101 has four lobes and the female rotor 102 has six lobes.

The gas to be compressed, normally air, is delivered to a working space of the compressor through an inlet port 108 and is then compressed in V-shaped working chambers formed between the rotors and the walls of the working space. Each working chamber moves to the right in figure 1 as the rotors 101, 102 rotate. The volume of a working chamber will thus decrease continuously during the latter part of its cycle, subsequent to communication with the inlet port having been cut off. The gas is thereby compressed and exits in a compressed state from the compressor through an outlet port 109. The ratio between outlet pressure and inlet pressure is determined by the inherent volumetric relationship between the volume of a working chamber immediately after its communication with the inlet port 108 has been cut off, and the volume of said chamber when it begins to communicate with the outlet port 109.

Figure 3 illustrates a compressor K, preferably a helical screw compressor, which is driven by a motor M via a shaft or axle 1. The compressor includes an inlet port 6 into which an inlet line 2 opens. The line 2 includes a check valve 3 which allows air to enter the compressor, while preventing the flow of air in the opposite direction. The compressor has at its other end an outlet port 7 which is connected to a pressure tank T via a line 4. One or more tools V, driven by compressed air, are supplied with pressure from the tank T via a line 5. The tank is provided with a pressure sensor 9 which is connected via a signal transmitting line 10 to a control means 8 that functions to control starting and stopping of the motor.

The pressure in the tank T shall vary between a highest pressure P1 and a lowest pressure P2. The motor M drives the compressor K until the pressure in the tank has reached said highest pressure P1, whereupon the motor M is switched off. When the pressure in the tank T has fallen to the lowest pressure P2, the motor M is restarted and again drives the compressor and therewith deliver compressed air to the tank T. The check valve 3 prevents compressed air from flowing from the tank T back through the compressor K and the inlet line 2.

Figure 4 illustrates diagrammatically a torque curve as a function of the rotational speed of an asynchronous motor. The axes are not graduated. The motor has a speed of N_4 for a torque of M_{2A}. When the torque of the motor increases to M_{1A}, the motor speed will drop to N_3. The relationship with respect to this asynchronous motor is at least substantially linear in one working range of said motor. The asynchronous motor thus has the property whereby a relatively large torque increase $\Delta M_k = (M_{1A} - M_{2A})$ leads to a relatively small reduction in motor speed.

As a result of this property of the asynchronous motor, the motor will be started when the tank pressure has fallen to the pressure P2, wherewith the compressor begins to compress air. Because of the small increase in speed required to raise the motor torque from M_{2A} to M_{1A}, the compressor will work at almost maximum capacity in this torque range. This results in a rapid increase in tank pressure. A compressor driven by an asynchronous motor will thus result in a short compressor operating time in achieving the desired highest pressure in the tank T. Only a relatively small volume of air responsible in lowering the tank pressure will be consumed during this relatively short period of time. This will result in frequent starting of the motor, in order to keep the tank pressure within the desired pressure range. These moments of frequent starting and stopping of the motor will significantly shorten its useful life, for instance as a result of overheating of the motor windings.

Similar to figure 4, figure 5 illustrates diagrammatically a torque curve as a function of motor speed. The illustrated curve of figure 5 relates to a commutator motor. The axes shown in figure 5 are not graduated. The torques M_{1k} and M_{2k} in figure 5 correspond to the torques M_{1A} and M_{2A} in figure 4. The commutator motor has a speed of N2 in respect of torqueM_{2k}. When the torque of said motor has increased to M_{1K}, the rpm of the motor will have fallen to N1. This relationship is at least substantially linear for the commutator in the working range. In the case of this motor, a relatively large increase in torque $\Delta M_k = (M_{1A} - M_{2A})$ will result in a relatively large reduction in motor speed.

As a result of this property of the commutator motor the tank pressure will have fallen to P_{2k} when the motor is started (see fig. 3) wherewith the compressor begins to compress air. Due to the significant increase in rpm. or motor speed, necessary for increasing the motor torque from M_{2k} to M_{1k}, it is necessary for the compressor to work over a significantly longer period of time to achieve maximum pressure than that required by an asynchronous motor. This means that it will take far longer to achieve a tank pressure P1 when the compressor is driven by a commutator motor. During this longer compressor working time the volume of air consumed is much greater than when a compressor is driven by an asynchronous motor, with which the maximum tank pressure is reached much more quickly. Thus, the number of starts involved when using a commutator motor is far less than the number of starts involved when driving the same compressor with an asynchronous motor in order to maintain the tank T pressurised.

According to one preferred embodiment of the invention there is used a compressor that has a relatively low internal volume factor. By internal volume factor is meant the relationship between the minimum and maximum thread volume enclosed in the helical rotor compressor used. The internal volume factor will preferably be such that the pressure of the compressor K will be less than P2 + 0.85 * (P1-P2) when the thread volume of the working chamber that commences communication with the tank T has its minimum volume. This means that the compressor outlet pressure in given working chamber will be at most equal to the lowest pressure of the tank plus 85 percent of the difference between the highest and the lowest pressure of the tank. The compressor will preferably be optimised for an internal volume factor at which the compressor pressure at the opening instance will be equal to the lowest working pressure P2 in the pressure container. It is particularly preferred that the compressor is optimised in respect of an internal volume factor at which the compressor pressure at the opening instance is lower than the lowest working pressure P2 in the pressure container.